CHAPTER ONE
Mechanical Drives

by

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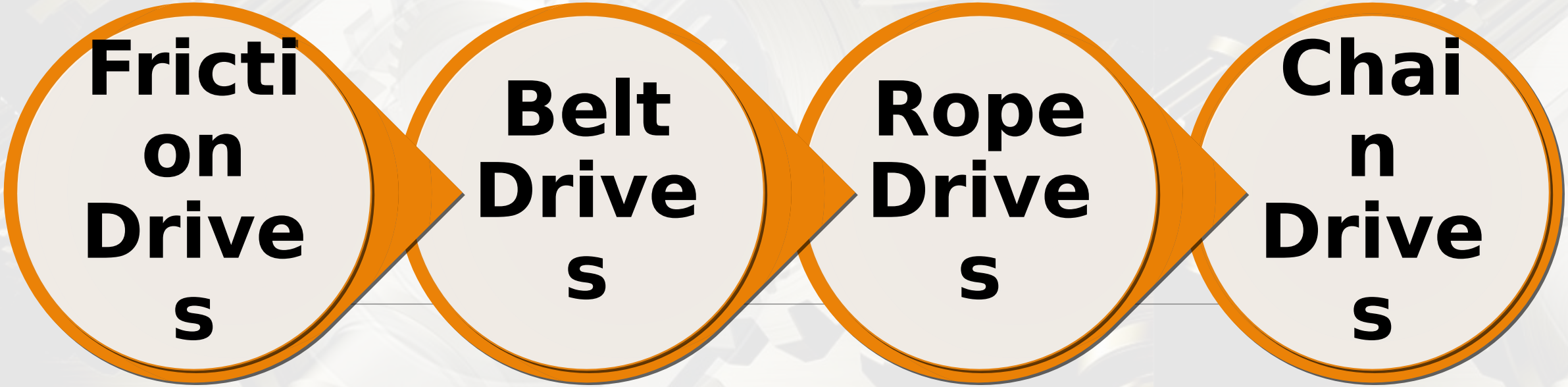
March
15, 2023

CONTENTS



Friction Drives

- A mechanical drive in which motion is transmitted or converted by the frictional forces between rolling bodies—cylinders, cones, and the like—that are pressed against each other.
- Friction drives may be used to transmit motion between shafts having parallel or transverse axes, to convert rotary motion to helical motion, and to convert rotary motion to translatory motion.
- They may have fixed or variable gear ratios.

Friction drives advantages

- calm and almost noiseless operation
- the drive can also work as a clutch
- it is possible to change rotations while the machine is running
- immediate equalization when the friction wheel slips

The friction drive disadvantages

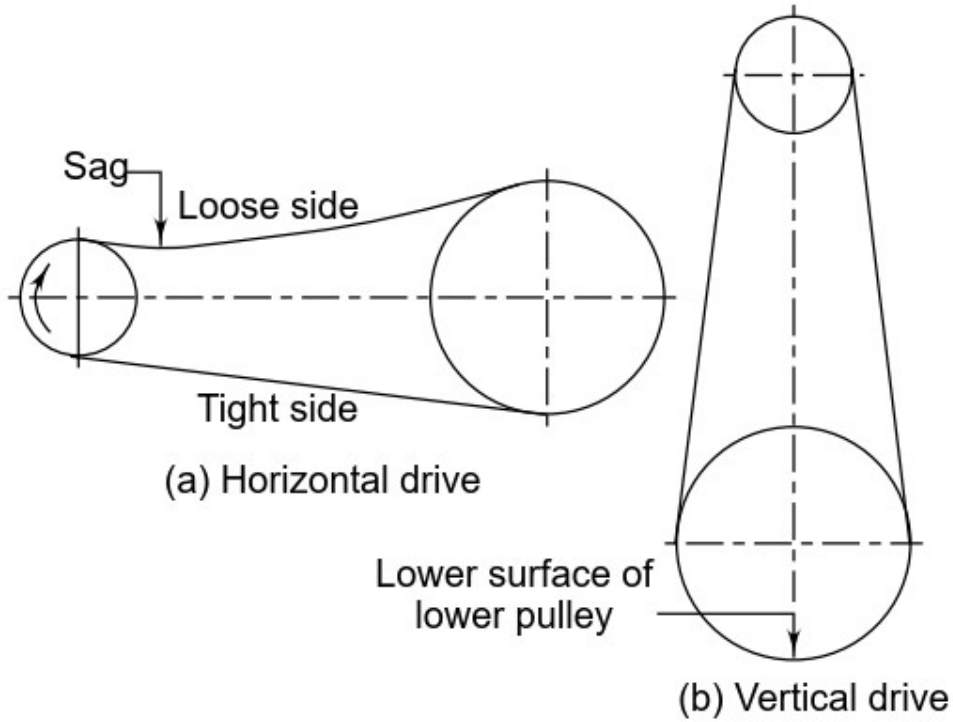
- Pressure on the shaft and bearings
- Instability of the gear speed ratio

Belt Drives

- A belt is a flexible power transmission element that seats tightly on a set of pulleys or sheaves.
- The belt is designed to ride around the two sheaves without slipping.
- The belt is installed by placing it around the two sheaves while the center distance between them is reduced. Then the sheaves are moved apart, placing the belt in a rather high initial tension.
- When the belt is transmitting power, friction causes the belt to grip the driving sheave, increasing the tension in one side, called the "tight side," of the drive.

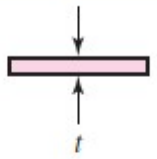
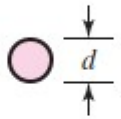
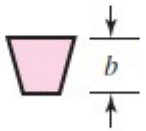
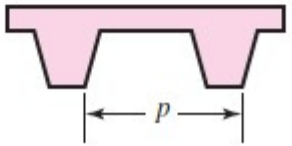
Cont …

- The tensile force in the belt exerts a tangential force on the driven sheave, and thus a torque is applied to the driven shaft.
- The opposite side of the belt is still under tension, ue. Thus, it is called the “slack



(i) Vertical belt drive should be avoided.
(ii) In horizontal belt drive, the upper side should be loose side.

Types of Belts

Belt Type	Figure	Joint	Size Range	Center Distance
Flat		Yes	t 0.03 to 0.20 in 0.75 to 5 mm	No upper limit
Round		Yes	d $\frac{1}{8}$ to $\frac{3}{4}$ in	No upper limit
V		None	b 0.31 to 0.91 in 8 to 19 mm	Limited
Timing		None	p 2 mm and up	Limited

- Crowned pulleys are used for flat belts, and grooved pulleys, or sheaves, for round and V belts. Timing belts require toothed wheels, or sprockets.

Advantages of Belts

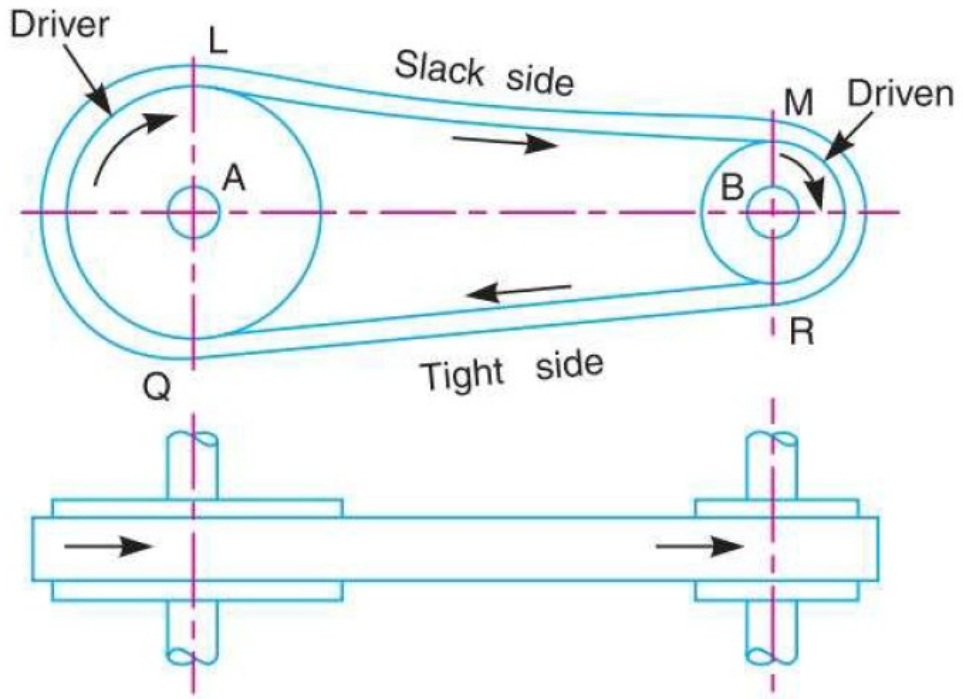
- Belt drives can transmit power over considerable distance between the axes of driving and driven shafts.
- The operation of belt drive is smooth and silent.
- They can transmit only a definite load, which if exceeded, will cause the belt to slip over the pulley, thus protecting the parts of the drive against overload.
- They have the ability to absorb the shocks and damp vibration.
- They are simple to design.
- They have low initial cost.

Disadvantages of Belts

- Belt drives have large dimensions and occupy more space.
- The velocity ratio is not constant due to belt slip.
- They impose heavy loads on shafts and bearings.
- There is considerable loss of power resulting in low efficiency.
- Belt drives have comparatively short service life.

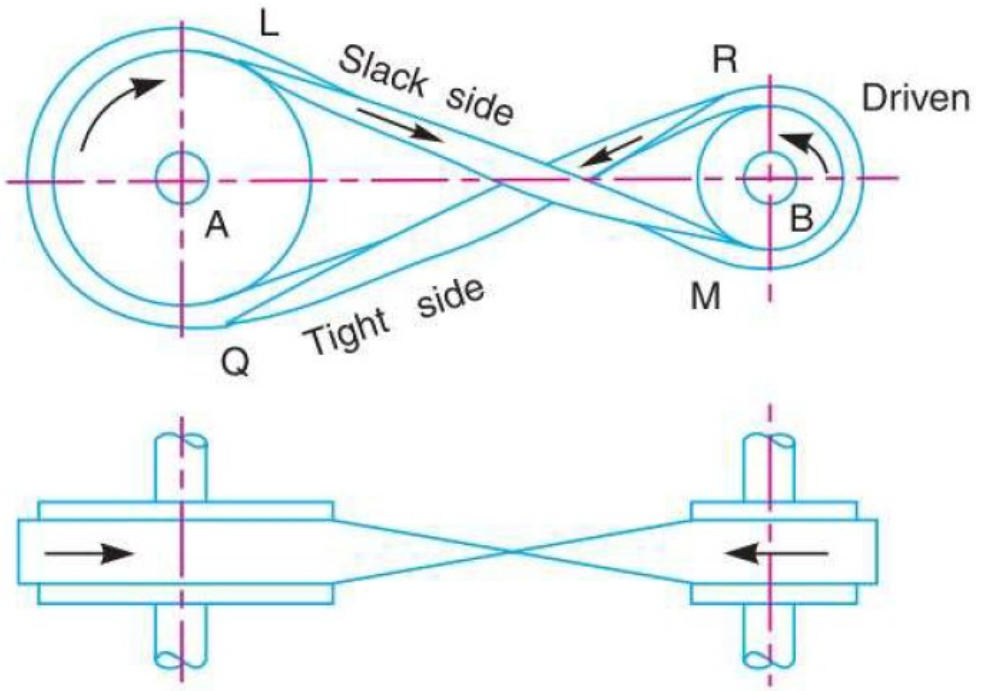
Belt Drive type

- Open Belt Drive: The open belt drive is used with shafts arranged parallel and rotating in the same direction



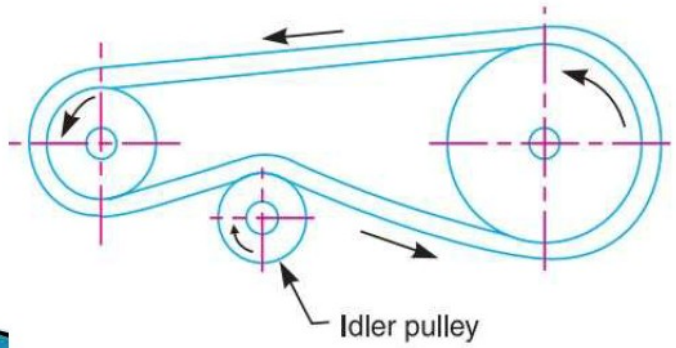
Belt Drive type

- Crossed Belt Drive: The crossed belt drive is used with shafts arranged parallel and rotating in the opposite direction

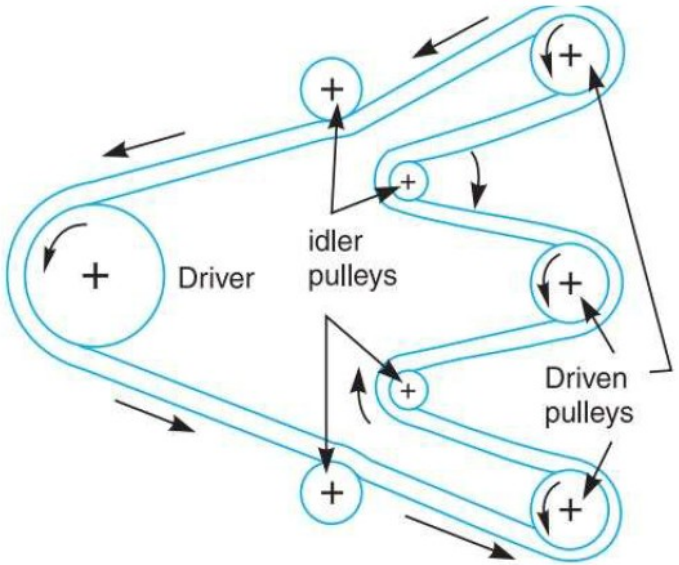


Belt Drive type

- **Belt Drive with Idler pulleys**: is used with shafts arranged parallel and when belt drive cannot be used due to small angle of contact on the smaller pulley.
- **Idler pulley** is a pulley which does not transmit power but assist power transmission by increasing angle of contact of smaller pulley

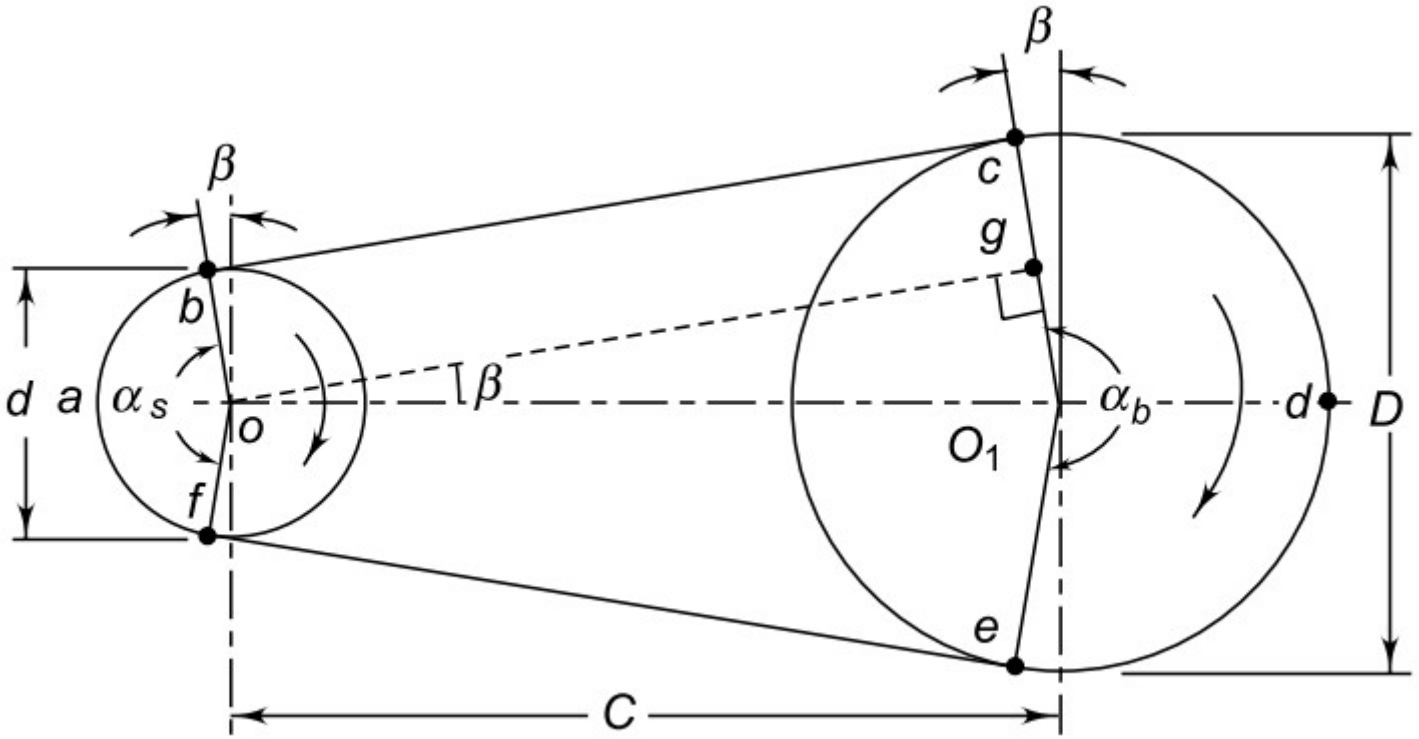


(*a*) Belt drive with single idler pulley.



(*b*) Belt drive with many idler pulleys.

Length of Belt (Open Belt Drive)



$$\theta = \sin^{-1}\frac{D-d}{2C}$$

$$\theta_s = 180 - 2\sin^{-1}\frac{D-d}{2C}$$

$$\theta_b = 180 + 2\sin^{-1}\frac{D-d}{2C}$$

a_s = wrap angle for small pulley (degrees)
a_b = wrap angle for big pulley (degrees)
D = diameter of big pulley (mm)
d = diameter of small pulley (mm)
C = centre distance (mm)

Length of Belt (Open Belt Drive)

- The length of the belt is given by:

$$L = \text{arc}\,(fab) + \overline{bc} + \text{arc}\,(cde) + \overline{ef}$$

$$= \frac{d}{2}(\alpha_s) + \overline{og} + \frac{D}{2}(\alpha_b) + \overline{og}$$

$$= \frac{d}{2}(\pi - 2\beta) + C\cos\beta + \frac{D}{2}(\pi + 2\beta) + C\cos\beta$$

or $$L = \frac{\pi(D+d)}{2} + \beta(D-d) + 2C\cos\beta \quad \text{.....(i)}$$

For small values of β,

$$\beta = \sin\beta = \left(\frac{D-d}{2C}\right) \quad \text{..... (a)}$$

and $$\cos\beta = 1 - 2\sin^2\left(\frac{\beta}{2}\right) = 1 - \frac{\beta^2}{2}$$

$$= 1 - \frac{(D-d)^2}{8C^2} \quad \text{..... (b)}$$

Length of Belt (Open Belt Drive)

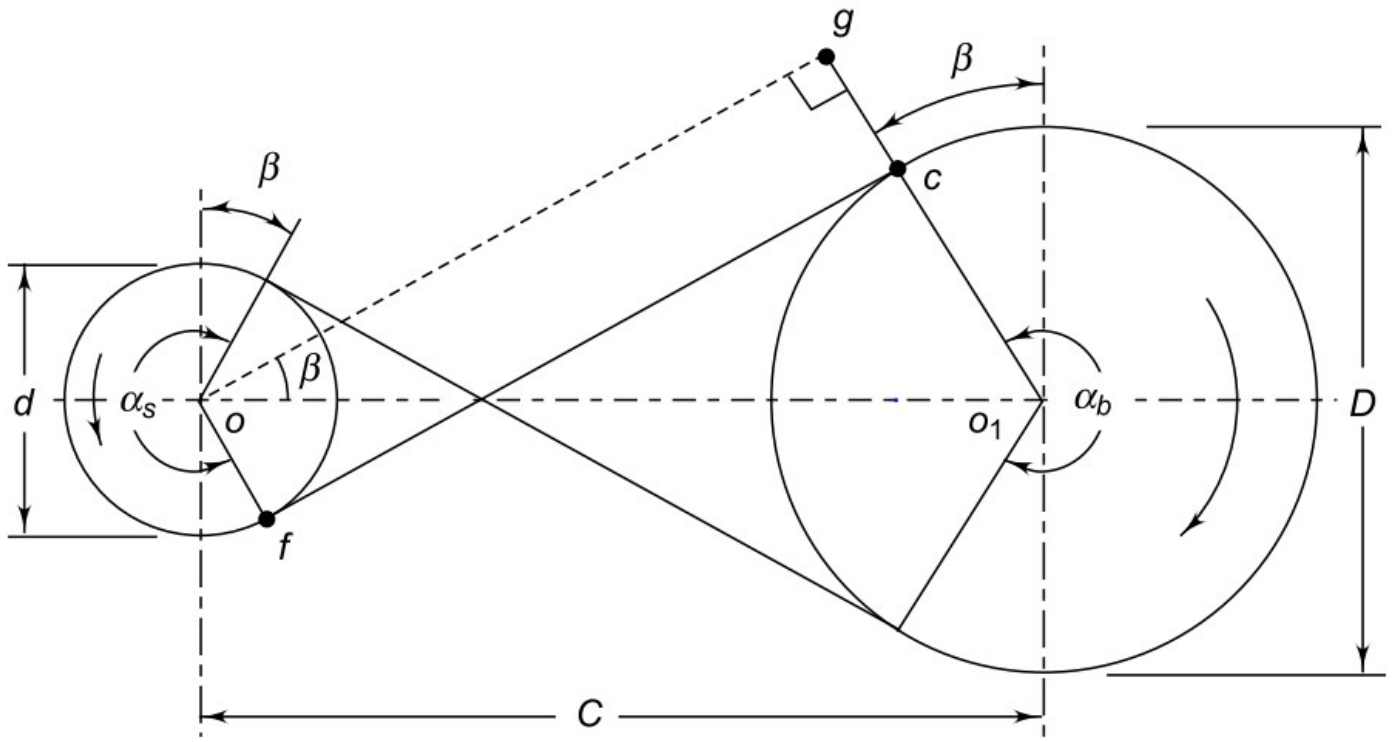
- Substituting eq. (a) and (b) into eq. (i)

$$L = \frac{\pi(D+d)}{2} + \left(\frac{D-d}{2C}\right)(D-d) + 2C\left[1 - \frac{(D-d)^2}{8C^2}\right]$$

$$= \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{2C} + 2C - \frac{(D-d)^2}{4C}$$

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

Length of Belt (Crossed Belt



$$\theta = \sin^{-1}\frac{D+d}{2C}$$

$$\theta_s = 180 + 2\sin^{-1}\frac{D+d}{2C}$$

$$\theta_b = 180 + 2\sin^{-1}\frac{D+d}{2C}$$

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D+d)^2}{4C}$$

ANALYSIS OF FLAT BELT TENSIONS

- The forces acting on the element of a flat belt are shown

P_1 = belt tension in the tight side (N)
P_2 = belt tension in the loose side (N)
m = mass of the one meter length of belt (kg/m)
v = belt velocity (m/s)
f = coefficient of friction
α = angle of wrap for belt (radians)

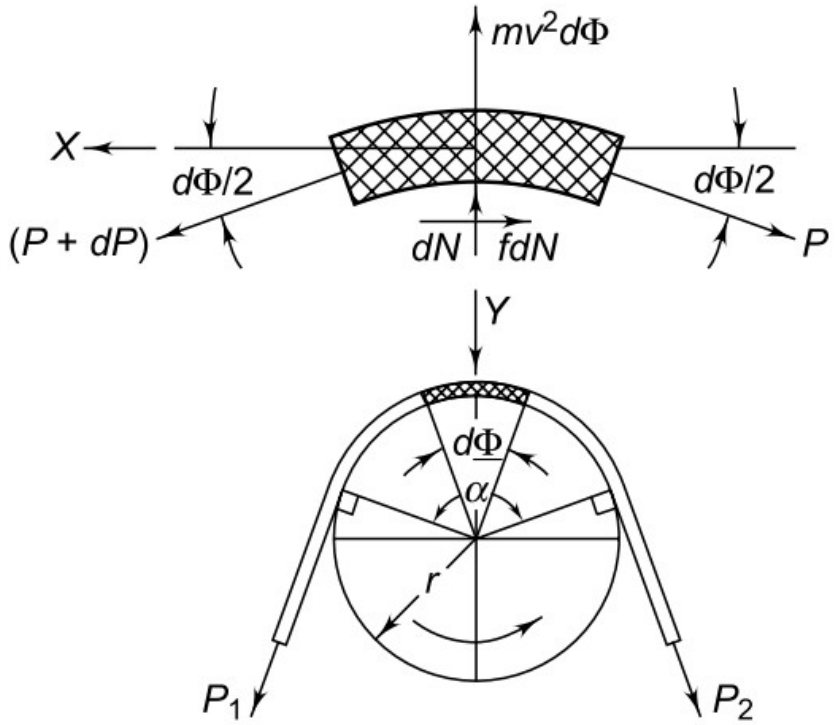


Fig. *Forces on Flat Belt*

Considering equilibrium of forces in X and Y directions,

$$(P + dP)\cos\left(\frac{d\phi}{2}\right) - P\cos\left(\frac{d\phi}{2}\right) - fdN = 0 \qquad \text{(a)}$$

$$(P + dP)\sin\left(\frac{d\phi}{2}\right) + P\sin\left(\frac{d\phi}{2}\right) - mv^2 d\phi - dN = 0 \qquad \text{(b)}$$

For small values of $\left(\frac{d\phi}{2}\right)$,

$\cos\left(\frac{d\phi}{2}\right)$ is approximately 1

$\sin\left(\frac{d\phi}{2}\right)$ is approximately $\left(\frac{d\phi}{2}\right)$

Substituting these values in Eq. (a),

$$(P + dP) - P - f\,dN = 0$$

or
$$dP - f\,dN = 0$$

$\therefore$
$$dN = \frac{dP}{f} \qquad \text{(c)}$$

Similarly, substituting $\sin\left(\frac{d\phi}{2}\right)$ as $\left(\frac{d\phi}{2}\right)$ in Eq. (b),

$$(P+dP)\left(\frac{d\phi}{2}\right)+P\left(\frac{d\phi}{2}\right)-mv^2 d\phi-dN=0$$

Neglecting the differential of the second order $(dP\times d\phi\,(dP\times d\phi)$,

$$P d\phi - mv^2\, d\phi - dN = 0$$

Substituting Eq. (c) in the above expression,

$$(P-mv^2)d\phi-\frac{dP}{f}=0$$

or
$$\frac{dP}{(P-mv^2)}=f d\phi$$

Integrating the above expression,

$$\int_{P_2}^{p_1}\frac{dP}{(P-mv^2)}=f\int_0^{\alpha}d\phi$$

$$[\log_e(P-mv^2)]_{P_2}^{p_1}=f[\phi]_0^{\alpha}$$

$$\log_e(P_1-mv^2)-\log_e(P_2-mv^2)=f(\alpha-0)$$

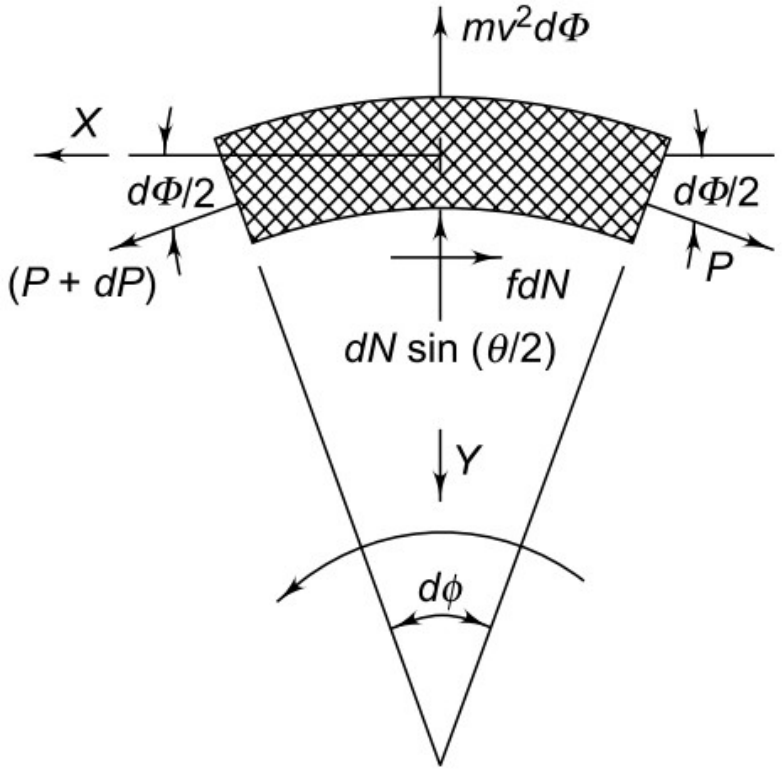
$$\log_e\left[\frac{P_1-mv^2}{P_2-mv^2}\right]=f\alpha$$

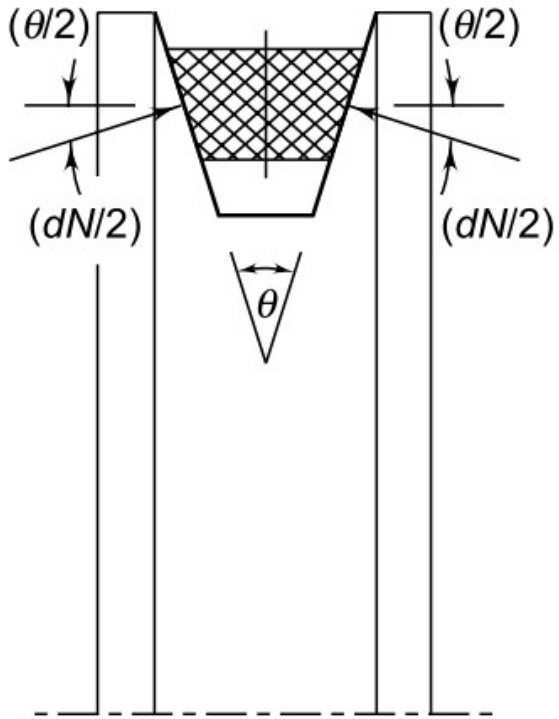
Therefore, the relationship is given by

$$\frac{P_1-mv^2}{P_2-mv^2}=e^{f\alpha} \qquad (13.6)$$

ANALYSIS OF V BELT TENSIONS

- The forces acting on the element of a V belt are shown below. The force components P, (P+dP), and are the same as those of the flat belt. The difference lies in the normal reaction dN. The normal reaction, which acts on two sides of the V-belt, is assumed as on each side.





$$(P + dP)\cos\left(\frac{d\phi}{2}\right) - P\cos\left(\frac{d\phi}{2}\right) - f\,dN = 0 \quad \text{(d)}$$

$$(P + dP)\sin\left(\frac{d\phi}{2}\right) + P\sin\left(\frac{d\phi}{2}\right)$$

$$-(m\,v^2 d\phi) - dN\sin\left(\frac{\theta}{2}\right) = 0 \quad \text{(e)}$$

For small values of $\left(\frac{d\phi}{2}\right)$,

$\cos\left(\frac{d\phi}{2}\right)$ is approximately 1

$\sin\left(\frac{d\phi}{2}\right)$ is approximately $\left(\frac{d\phi}{2}\right)$

Substituting these values in Eq. (d),

$$(P + dP) - P - f\,dN = 0$$

or $$dP - f dN = 0$$

$$\therefore \quad dN = \frac{dP}{f} \quad \text{(f)}$$

Similarly, substituting $\sin\left(\frac{d\phi}{2}\right)$ as $\left(\frac{\theta}{2}\right)$ in Eq. (e),

$$(P + dP)\left(\frac{d\phi}{2}\right) + P\left(\frac{d\phi}{2}\right) - mv^2 d\phi$$

$$-dN\sin\left(\frac{\theta}{2}\right) = 0$$

Neglecting the term of the second order differential ($dP \times d\phi$),

$$P d\phi - mv^2 d\phi - dN\sin\left(\frac{\theta}{2}\right) = 0$$

Substituting Eq. (f) in the above expression,

$$(P - mv^2)\,d\phi - \frac{dP}{f}\sin\left(\frac{\theta}{2}\right) = 0$$

or $$\frac{dP}{(P - mv^2)} = \frac{f d\phi}{\sin\left(\frac{\theta}{2}\right)}$$

Integrating the above expression,

$$\int_{P_2}^{p_1} \frac{dP}{(P - mv^2)} = \frac{f}{\sin\left(\frac{\theta}{2}\right)} \int_0^{\alpha} d\phi$$

$$[\log_e (P - mv^2)]_{P_2}^{p_1} = \frac{f}{\sin\left(\frac{\theta}{2}\right)} \times [\phi]_0^{\alpha}$$

Substituting limits,

$$\log_e (P_1 - mv^2) - \log_e (P_2 - mv^2) = \frac{f}{\sin\left(\frac{\theta}{2}\right)} (\alpha - 0)$$

$$\log_e \left[\frac{P_1 - mv^2}{P_2 - mv^2}\right] = \frac{f\alpha}{\sin\left(\frac{\theta}{2}\right)}$$

or

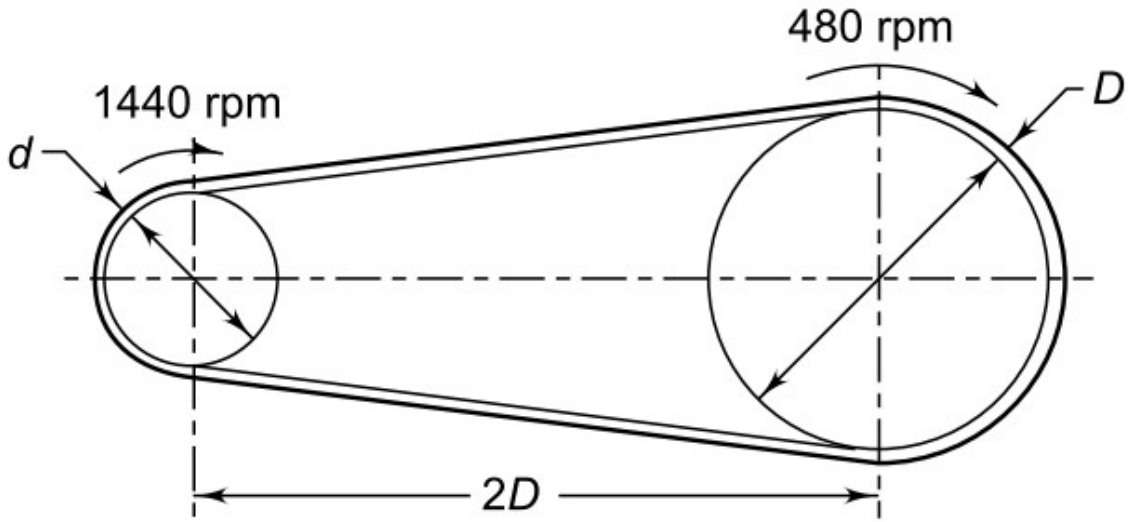
$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha / \sin(\theta/2)} \qquad (13.7)$$

$$\text{Power transmitted} = (P_1 - P_2)\, v \quad \text{N-m/s or W}$$

or
$$\text{kW} = \frac{(P_1 - P_2)v}{1000} \qquad (13.8)$$

Example

- The layout of a leather belt drive transmitting 15 kW of power is shown in below. The centre distance between the pulleys is twice the diameter of the bigger pulley. The belt should operate at a velocity of 20 m/s approximately and the stresses in the belt should not exceed 2.25 N/ . The density of leather is 0.95 g/ and the coefficient of friction is 0.35. The thickness of the belt is 5 mm. Calculate:
  i. The diameter of pulleys;
  ii. The length and width of the belt; and
  iii.



Solution

Given kW = 15 $v = 20$ m/s $C = 2D$
$t = 5$ mm $\rho = 0.95$ g/cc $\sigma = 2.25$ N/mm^2
$f = 0.35$

Step I *Diameter of pulleys*

$$v = \frac{\pi dn}{60(1000)}$$

or
$$d = \frac{60(1000)v}{\pi n} = \frac{60(1000)(20)}{\pi(1440)}$$
$$= 265.26 \text{ mm (or 270 mm)}$$

The diameters of pulleys are

$$d = 270 \text{ mm}$$

and
$$D = \frac{270(1440)}{(480)} = 810 \text{ mm} \qquad \text{(a)}$$

Step II *Belt length*

$$C = 2D = 2(810) = 1620 \text{ mm}$$

From Eq. (13.3),

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$
$$= 2(1620) + \frac{\pi(810+270)}{2} + \frac{(810-270)^2}{4(1620)}$$
$$= 4981.46 \text{ mm} \qquad \text{(b1)}$$

Step III *Belt width and belt tensions*

The correct belt velocity is given by,

$$v = \frac{\pi dn}{60(1000)} = \frac{\pi(270)(1440)}{60(1000)} = 20.36\,\text{m/s}$$

$$\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

$$= 180 - 2\sin^{-1}\left(\frac{810-270}{2\times 1620}\right) = 160.8°$$

or $$\alpha_s = \left(\frac{160.8}{180}\right)\pi = 2.81\,\text{rad}$$

The density of leather is given as 0.95 g per cubic cm. The volume of 1 metre belt in cubic cm is given by,

$$\text{volume} = (\text{length}) \times (\text{width}) \times (\text{thickness})$$

$$= (100)\left(\frac{b}{10}\right)\left(\frac{5}{10}\right)$$

where b is the width in mm.

$$\therefore \quad m = (0.95)(100)\left(\frac{b}{10}\right)\left(\frac{5}{10}\right)\text{ g/m}$$

$$= (0.95)\,(100)\left(\frac{b}{10}\right)\left(\frac{5}{10}\right)(10^{-3})\text{ kg/m}$$

$$= (4.75\times 10^{-3})b\text{ kg/m}$$

or $$mv^2 = (4.75\times 10^{-3})b\,(20.36)^2 = 1.97b$$

also $$e^{f\alpha} = e^{(0.35)(2.81)} = 2.67$$

$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha} \quad \text{or} \quad \frac{P_1 - 1.97b}{P_2 - 1.97b} = 2.67$$

or $$P_1 - 2.67P_2 + 3.29b = 0 \qquad \text{(i)}$$

The maximum stress in the belt is given as 2.25 N/mm^2.

$$\sigma = \frac{\text{maximum tension}}{\text{area of cross-section of belt}} = \frac{P_1}{A}$$

$$\therefore \qquad P_1 = \sigma A = 2.25(5b) = (11.25\ b)\ \text{N} \qquad \text{(ii)}$$

From Eq. (13.8),

$$P_1 - P_2 = \frac{1000(\text{kW})}{v} = \frac{1000(15)}{20.36} = 736.74\ \text{N} \qquad \text{(iii)}$$

Solving Eqs (i), (ii) and (iii) simultaneously, we have

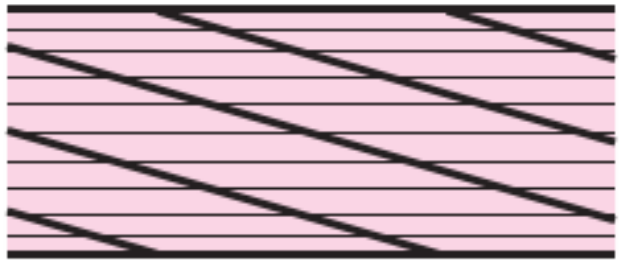
$$b = 127.02\ \text{mm or } 130\ \text{mm} \qquad \text{(b2)}$$

$$P_1 = 1428.98\ \text{N} \quad \text{and} \quad P_2 = 692.26\ \text{N} \qquad \text{(c)}$$

Rope Drive

- **Wire rope** is a flexible rope constructed by laying steel wires into various patterns of multi-wired strands around a core system to produce a helically wound rope.
- Wire rope is made with two types of winding.

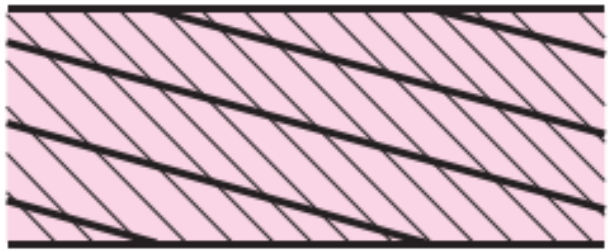
1. **The regular lay**, which is the accepted standard, has the wire twisted in one direction to form the strands, and the strands twisted in the opposite direction to form the rope.
   - In the completed rope the visible wires are approximately parallel to the axis of the rope.
   - Regular-lay ropes do not kink or untwist and are easy to handle.



(*a*) Regular lay

Cont...

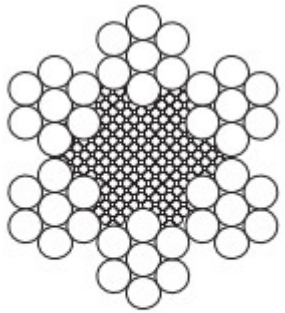
2. **Lang-lay ropes** have the wires in the strand and the strands in the rope twisted in the same direction, and hence the outer wires run diagonally across the axis of the rope.
   - Lang-lay ropes are more resistant to abrasive wear and failure due to fatigue than are regular-lay ropes, but they are more likely to kink and untwist.



(*b*) Lang lay

Wire rope designation

- Wire rope is designated as, for example, a 1 -in 6 × 7 haulage rope.
  - The first figure is the diameter of the rope
  - The second and third figures are the number of strands and the number of wires in each strand, respectively.



(*c*) Section of 6 × 7 rope

The stress in one of the wires of a rope passing around a sheave may be calculated as follows. From solid mechanics, we have

$$M = \frac{EI}{\rho} \qquad \text{and} \qquad M = \frac{\sigma I}{c}$$

where the quantities have their usual meaning. Eliminating M

$$\sigma = \frac{Ec}{\rho}$$

For the radius of curvature ρ, we can substitute the sheave radius $D/2$. Also, $c = d_w/2$, where d_w is the wire diameter. These substitutions give

$$\sigma = E_r \frac{d_w}{D}$$

where E_r is the *modulus of elasticity of the rope*, not the wire.

A wire rope tension giving the same tensile stress as the sheave bending is called the *equivalent bending load* F_b, given by

$$F_b = \sigma A_m = \frac{E_r d_w A_m}{D}$$

The area of the metal in standard hoisting and haulage rope is $A_m = 0.38d^2$

- A wire rope may fail because the static load exceeds the ultimate strength of the rope. The first consideration in selecting a wire rope is to determine the static load. This load is composed of the following items:
  - ✓ The known or dead weight
  - ✓ Additional loads caused by sudden stops or starts
  - ✓ Shock loads
  - ✓ Sheave-bearing friction

The factor of safety is defined as:

$$n = \frac{F_u}{F_t}$$

where F_u is the ultimate wire load and F_t is the largest working tension.

- For an average operation, use a factor of safety of 5. Factors of safety up to 8 or 9 are used if there is danger to human life and for very critical situations.

- Once you have made a tentative selection of a rope based upon static strength, the next consideration is to ensure that the wear life of the rope and the sheave or sheaves meets certain requirements. When a loaded rope is bent over a sheave, the rope stretches like a spring, rubs against the sheave, and causes wear of both the rope and the sheave.

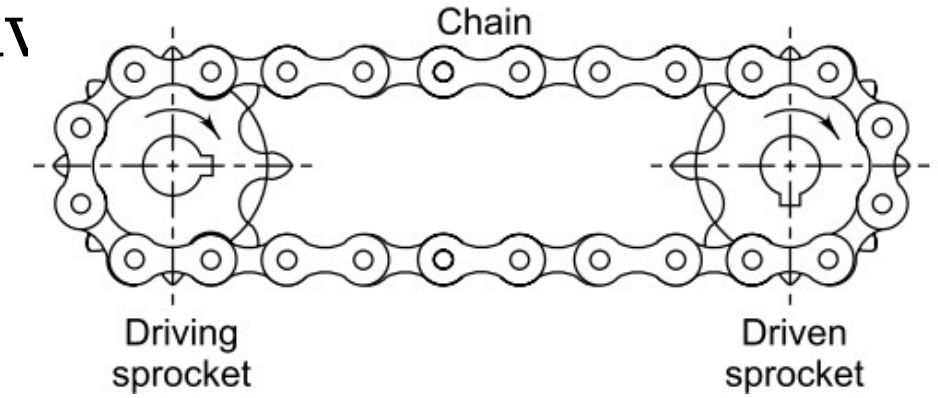
- The amount of wear that occurs depends upon the pressure of the rope in the sheave groove. This pressure is called the bearing pressure; a good estimate of its magnitude is given by:

$$p = \frac{2F}{dD}$$

where F = tensile force on rope
d = rope diameter
D = sheave diameter

Chain Drives

- A chain drive is a mechanical operating system where we use different types of chains to transmit power.
- A chain can be defined as a series of links connected by pin joints.
- The chain driv n belt and gear drives.



- With respect to their purpose, chains are classified into the following three groups:

A. Power transmission chains are used for transmitting power

Chain Drives cont…

B. Load lifting chains are used for suspending, raising or lowering loads in materials handling equipment.

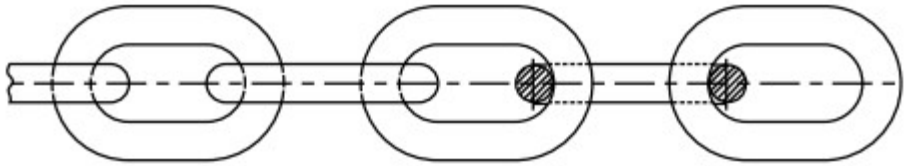


Fig. *Link Chain*

C. Hauling chains are used for carrying materials continuously by sliding, pulling or carrying in conveyors.

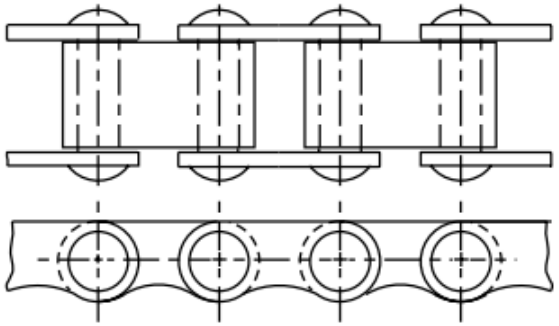
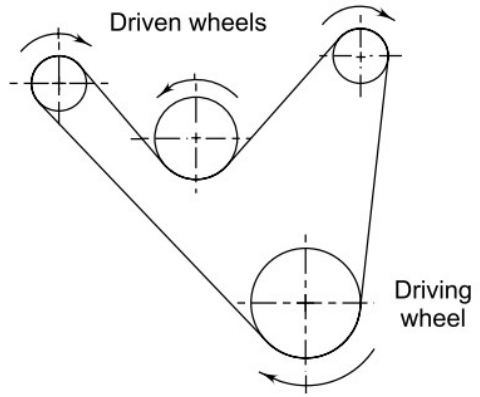


Fig. *Block Chain*

Advantages of Chain Drives

- Chain drives can be used for long as well as short centre distances.
- A number of shafts can be driven in the same or opposite direction by means of [illegible] from a single driving sprocket.



- Chain drives have small overall dimensions than belt drives, resulting in compact unit.
- A chain does not slip and to that extent, chain drive is a positive drive

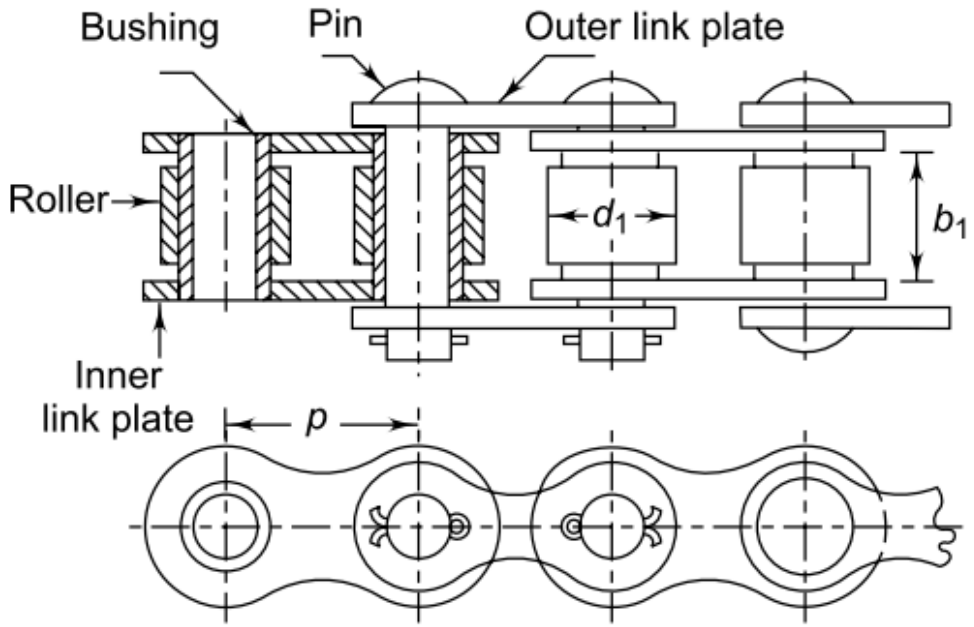
Disadvantages of Chain Drives

A. Chain drives operate without full lubricant film between the joints unlike gears. This results in more wear at the joints.
B. Chain drives are not suitable for non-parallel shafts.
C. Chain drive is unsuitable where precise motion is required due to polygonal effect.
D. Chain drives require housing.
E. Compared with belt drives, chain drives require precise alignment of shafts.
F. Chain drives require adjustment for slack, such as a tensioning device.
G. Chain drives generate noise.

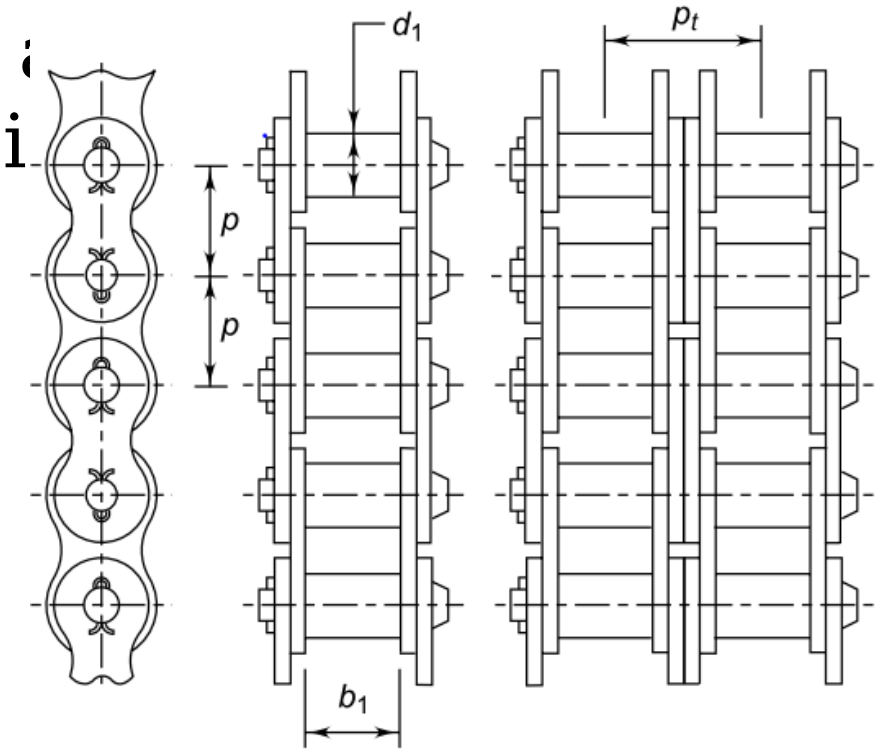
Roller Chains

A roller chain consists of following five parts:

I. Pin
II. Bushing
III. Roller
IV. Inner link plate
V. Outer link plate



- The inner and outer link plates are made of medium carbon steels. These link plates are blanked from cold-rolled sheets and hardened to 50 HRC.
- The pins, bushes and rollers are made of case carburising alloy steels and hardened to 50 HRC.
- The pitch (p) of the chain is the linear distance between the a… The roller chains are desi… tch”.

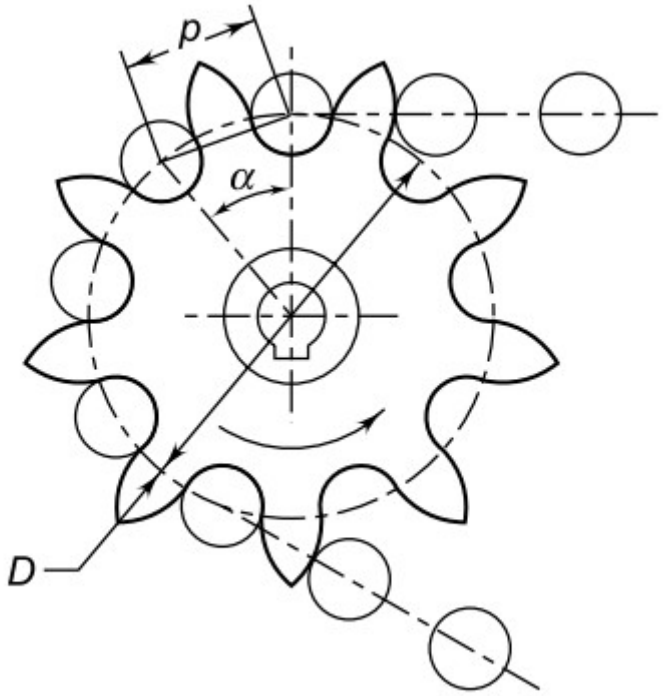


Simple Chain Duplex Chain

Single-strand – simple chain
Multi-strand constructions - (duplex or triplex chains)

GEOMETRIC RELATIONSHIPS

- The engagement of chain on sprocket wheel is shown in the following figure. D is the pitch circle diameter of the sprocket and is called the pitch angle.
- The pitch circle diameter of the sprocket is defined as the diameter of an imaginary circle that passes through the centres of link pins as the chain is wrapped or



$$\alpha = \frac{360}{z} \qquad (14.1)$$

where z is the number of teeth on the sprocket. From the figure, it can be proved that

$$\sin\left(\frac{\alpha}{2}\right) = \frac{(p/2)}{(D/2)} \quad \text{or} \quad D = \frac{p}{\sin\left(\frac{\alpha}{2}\right)}$$

$$\therefore \qquad D = \frac{p}{\sin\left(\frac{180}{z}\right)} \qquad (14.2)$$

The velocity ratio i of the chain drives is given by,

$$i = \frac{n_1}{n_2} = \frac{z_2}{z_1} \qquad (14.3)$$

where

n_1, n_2 = speeds of rotation of driving and driven shafts (rpm)

z_1, z_2 = number of teeth on driving and driven sprockets.

The average velocity of the chain is given by,

$$v = \frac{\pi D n}{60 \times 10^3}$$

$$\therefore \qquad v = \frac{zpn}{60 \times 10^3} \qquad (14.4)$$

where v is the average velocity in m/s.

The length of the chain is always expressed in terms of the number of links, or

$$L = L_n \times p \qquad (14.5)$$

where

L = length of the chain (mm)

L_n = number of links in the chain

The number of links in the chain is determined by the following approximate relationships:

$$L_n = 2\left(\frac{a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \times \left(\frac{p}{a}\right) \qquad (14.6)$$

where,

a = centre distance between axes of driving and driven sprockets (mm)

z_1 = number of teeth on the smaller sprocket

z_2 = number of teeth on the larger sprocket

- The number of links () is adjusted to the previous or next digit so as to get an even number. It is always preferred to have an ‘even’ number of links, since the chain consists of alternate pairs of inner and outer link plates.
- After selecting the exact number of links, the centre to centre distance betwe is calculated by the following fo

$$a = \frac{p}{4}\left\{\left[L_n - \left(\frac{z_1+z_2}{2}\right)\right] + \sqrt{\left[L_n - \left(\frac{z_1+z_2}{2}\right)\right]^2 - 8\left[\frac{z_2-z_1}{2\pi}\right]^2}\right.$$

POLYGON EFFECT

- **Polygon effect**: causes that the velocity ratio transmitted changes periodically, and the velocity of driven sprocket is variational, while the velocity of driving sprocket is constant

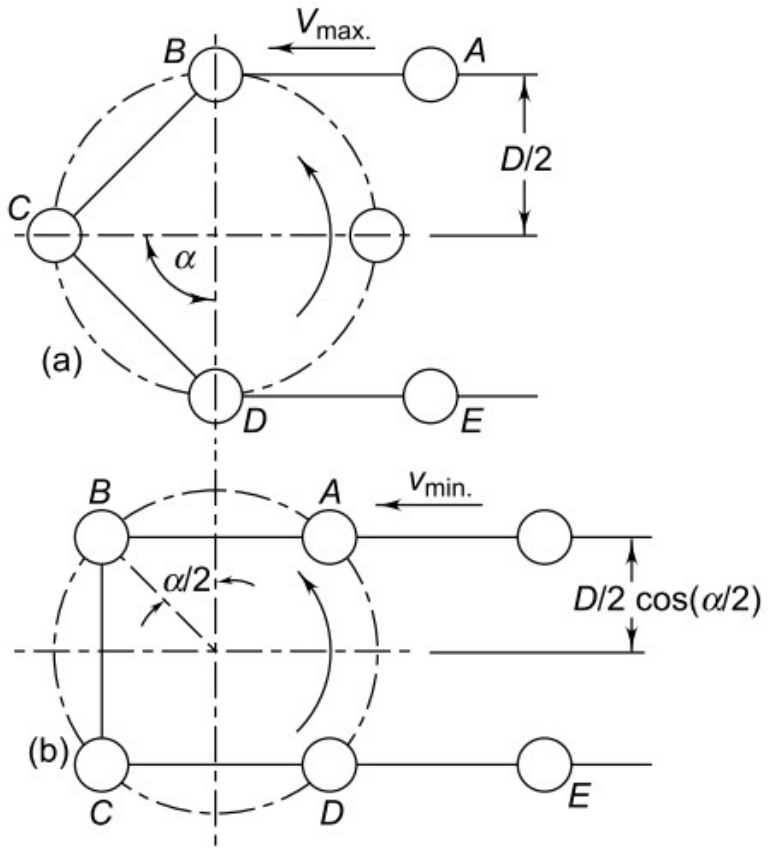


Fig. *Polygonal action of Chain*

$$v_{max.} = \frac{\pi Dn}{60 \times 10^3} \text{ m/s}$$

$$v_{min.} = \frac{\pi Dn \cos\left(\frac{\alpha}{2}\right)}{60 \times 10^3} \text{ m/s}$$

Cont…

- It is evident that the linear speed of the chain is not uniform but varies from to during every cycle of tooth engagement. This results in a pulsating and jerky motion. The variation in velocity is given by

$$(v_{max.} - v_{min.}) \propto \left[1 - \cos\left(\frac{\alpha}{2}\right)\right]$$

or

$$(v_{max.} - v_{min.}) \propto \left[1 - \cos\left(\frac{180}{z}\right)\right]$$

POWER RATING OF ROLLER CHAINS

- The power transmitted by the roller chain can be expressed by the elem

$$\text{kW} = \frac{P_1 v}{1000}$$

where

P_1 = allowable tension in the chain (N)

v = average velocity of chain (m/s)

- For a given application, the kW rating of the chain is determined by the following relationship:

$$\text{kW rating of c} = \frac{(\text{kW to be transmitted}) \times K_s}{K_1 \times K_2}$$

where

K_s = service factor

K_1 = multiple strand factor

K_2 = tooth correction factor

Table 14.1 *Dimensions and breaking loads of roller chains*

ISO chain number	*Pitch p (mm)*	*Roller diameter d_1 (mm) (max.)*	*Width b_1 (mm) (min.)*	*Transverse pitch p_t (mm)*	*Breaking load (min) N*		
					Simple	*Duplex*	*Triplex*
05B	8.00	5.00	3.00	5.64	4 400	7 800	11 100
06B	9.525	6.35	5.72	10.24	8 900	16 900	24 900
08A (ANSI-40)	12.70	7.95	7.85	14.38	13 800	27 600	41 400
08B	12.70	8.51	7.75	13.92	17 800	31 100	44 500
10A (ANSI-50)	15.875	10.16	9.4	18.11	21 800	43 600	65 400
10B	15.875	10.16	9.65	16.59	22 200	44 500	66 700
12A (ANSI-60)	19.05	11.91	12.57	22.78	31 100	62 300	93 400
12B	19.05	12.07	11.68	19.46	28 900	57 800	86 700
16A (ANSI-80)	25.40	15.88	15.75	29.29	55 600	111 200	166 800
16B	25.40	15.88	17.02	31.88	42 300	84 500	126 800
20A(ANSI-100)	31.75	19.05	18.90	35.76	86 700	173 500	260 200
20B	31.75	19.05	19.56	36.45	64 500	129 000	193 500
24A (ANSI-120)	38.10	22.23	25.22	45.44	124 600	249 100	373 700
24B	38.10	25.40	25.40	48.36	97 900	195 700	293 600
28A(ANSI-140)	44.45	25.40	25.22	48.87	169 000	338 100	507 100
28B	44.45	27.94	30.99	59.56	129 000	258 000	387 000
32A(ANSI-160)	50.80	28.58	31.55	58.55	222 400	444 800	667 200
32B	50.80	29.21	30.99	58.55	169 000	338 100	507 100
40A(ANSI-200)	63.50	39.68	37.85	71.55	347 000	693 900	1040 900
40B	63.50	39.37	38.10	72.29	262 400	524 900	787 300
48A	76.20	47.63	47.35	87.83	500 400	1000 800	1501 300
48B	76.20	48.26	45.72	91.21	400 300	800 700	1201 000
64B	101.60	63.50	60.96	119.89	711 700	1423 400	—

Table 14.2 *Power rating of simple roller chain*

Pinion speed (rpm)	*Power (kW)*								
	06 B	*08A*	*08 B*	*10A*	*10 B*	*12A*	*12 B*	*16A*	*16 B*
50	0.14	0.28	0.34	0.53	0.64	0.94	1.07	2.06	2.59
100	0.25	0.53	0.64	0.98	1.18	1.74	2.01	4.03	4.83
200	0.47	0.98	1.18	1.83	2.19	3.40	3.75	7.34	8.94
300	0.61	1.34	1.70	2.68	3.15	4.56	5.43	11.63	13.06
500	1.09	2.24	2.72	4.34	5.01	7.69	8.53	16.99	20.57
700	1.48	2.95	3.66	5.91	6.71	10.73	11.63	23.26	27.73
1000	2.03	3.94	5.09	8.05	8.97	14.32	15.65	28.63	34.89
1400	2.73	5.28	6.81	11.18	11.67	14.32	18.15	18.49	38.47
1800	3.44	6.98	8.10	8.05	13.03	10.44	19.85	—	—
2000	3.80	6.26	8.67	7.16	13.49	8.50	20.57	—	—

Table 14.3 *Service factor (K_s)*

Type of driven load	*Type of input power*		
	IC engine with hydraulic drive	*Electric motor*	*IC engine with mechanical drive*
(i) *Smooth:* agitator, fan, light conveyor	1.0	1.0	1.2
(ii) *Moderate shock*: machine tools, crane, heavy conveyor, food mixer, grinder	1.2	1.3	1.4
(iii) *Heavy shock:* punch press, hammer mill, reciprocating conveyor, rolling mill drive	1.4	1.4	1.7

Table 14.4 *Multiple strand factor (K_1)*

Number of strands	K_1
1	1.0
2	1.7
3	2.5
4	3.3
5	3.9
6	4.6

Table 14.5 *Tooth correction factor (K_2)*

Number of teeth on the driving sprocket	K_2
15	0.85
16	0.92
17	1.00
18	1.05
19	1.11
20	1.18
21	1.26
22	1.29
23	1.35
24	1.41
25	1.46
30	1.73

DESIGN OF CHAIN DRIVE

- There are two important rules in the design of a chain drive. They are as follows:
  I. The number of pitches or links of the chain should be always ‘even’.
  II. The number of teeth on the driving sprocket should be always ‘odd’, such as 17, 19 or 21.
- The odd number of teeth of the sprocket, in combination of even number of chain links, facilitates uniform wear.
- The minimum number of teeth on the driving sprocket is 17. From durability and noise considerations, the minimum number of teeth should be 21.
- When the drive operates at low speed such as 100 rpm, the number of teeth on the driving sprocket can be less than 17. In such cases, the number of teeth is taken as 13 or 15.

Example

- It is required to design a chain drive to connect 5 kW, 1400 rpm electric motor to a drilling machine. The speed reduction is 3 : 1. The centre distance should be approximately 500 mm.
  a. Select a proper roller chain for the drive.
  b. Determine the number of chain links.
  c. Specify the correct centre distance between the axes of sprockets.

Given kW = 5 $a = 500$ mm $n_1 = 1400$ rpm
$i = 3$

Step I *kW rating of chain*

The number of teeth on the driving sprocket is selected as 21. It is further assumed that the chain is a simple roller chain with only one strand. From Table 14.3, the service factor is taken as 1.3 assuming moderate shock conditions. For single strand chain,

$$K_1 = 1$$

For 21 teeth,

$$K_2 = 1.26$$

From Eq. (14.8),

$$\text{kW rating of chain} = \frac{(\text{kW to be transmitted}) \times K_s}{K_1 \times K_2}$$

$$= \frac{5 \times 1.3}{1 \times 1.26} = 5.16 \text{ kW}$$

Step II *Selection of chain*

Referring to Table 14.2, the power rating of the chain 8A at 1400 rpm is 5.28 kW. Therefore, the chain number 8A is selected.

Step III *Number of chain links*

The pitch dimension (p) of this chain (Table 14.1) is 12.70 mm. (i)

$z_1 = 21$ teeth $\quad z_2 = iz_1 = 3(21) = 63$ teeth

$p = 12.70$ mm $\quad a = 500$ mm

From Eq. (14.6),

$$L_n = 2\left(\frac{a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \times \left(\frac{p}{a}\right)$$

$$= 2\left(\frac{500}{12.70}\right) + \left(\frac{21+63}{2}\right) + \left(\frac{63-21}{2\pi}\right)^2 \times \left(\frac{12.70}{500}\right)$$

$$= 121.87 \text{ or } 122 \text{ links} \qquad \text{(ii)}$$

Step IV *Correct centre distance*

$$\left[L_n - \left(\frac{z_1 + z_2}{2}\right)\right] = \left[122 - \left(\frac{21+63}{2}\right)\right] = 80$$

From Eq. (14.7),

$$a = \frac{p}{4}\left\{\left[L_n - \left(\frac{z_1 + z_2}{2}\right)\right] + \sqrt{\left[L_n - \left(\frac{z_1 + z_2}{2}\right)\right]^2 - 8\left[\frac{z_2 - z_1}{2\pi}\right]^2}\right\}$$

$$= \frac{12.70}{4}\left\{80 + \sqrt{(80)^2 - 8\left[\frac{63-21}{2\pi}\right]^2}\right\}$$

$$= 500.81 \text{ mm} \qquad \text{(iii)}$$